HIGH VELOCITY IMPACT

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I ABSTRACT

A machine was designed and constructed that was intended to deliver an impact load of high energy on a tensile specimen at speeds varying from 100-500 ft. per second. Instrumentation, using resistance type strain gages was set up to measure and record strain versus time at two points on the sample during the period of rupture. One specimen was tested at 57 fps with this arrangement. Two other specimens were tested without instrumentation at 132 and 189 fps.

Elongation and reduction of area data were compared with static values. On highest velocity test an element of the machine failed mechanically and put an end to testing. Results indicated further modifications of machine and instrumentation are necessary before large-scale investigations can be undertaken. Results of tests on 2S aluminum showed increasing ductility and reduction of area with increased speed in region beyond critical speed contrary to usually accepted theory.

I N D E X

II OBJECT

1. To develop a method of dynamically loading a tensile specimen at velocities within the limits of 100 and 500 fps.

2. To determine the tensile properties of aluminum under high velocity impact conditions by means of an electrical strain gage directly attached to the specimen in the vicinity of fracture and by comparison of elongation and reduction of area at various velocities.

III INTRODUCTION

The ever-mounting demand for speed and more speed in the rotating and translatory devices of a modern technology that is reaching for the last fraction of increased efficiency has, in the last decade, confronted designers and engineers with the problem of designing for impact at high velocities. Admittedly the problem insofar as strength alone is concerned can be circumvented by liberal application of very arbitrary safety factors and this, in general, has been the rule for static structures. There are certain fields, however, in which the rewards accruing to a superior design that takes fullest advantage of the true impact strength of the materials in use are so rich that the principles of such design methods can no longer be overlooked. This is most true in applications where weight is an important factor and merely fattening up a section cannot replace a thorough and logical stress analysis. Ordnance equipment of various types, both projectiles and launching structures, aircraft elements, blast-proof designs, and applications in a yet unpredictable atomic energy technology, might be listed among the fields

that would be most affected by a complete exposition of the high velocity impact problem, resulting, it might reasonably be expected, in designs of higher efficiency and greater economy.

Prior to about 1936, very little was known of the behavior of materials under high velocity loading. At that time H. C. Mann carried out a series of tests[2] which opened the field and pointed up the prevailing ignorance of the fundamental theory underlying the phenomena observed. It may be said that the presentation of the subject as a legitimate analytical branch of materials theory came with the reading in 1941 of a paper by Theodore Von Karman before the National Academy of Science, "On the Propagation of Plastic Deformation in Solids"[11], in which was introduced an analysis and a theoretical basis for the behavior of materials under high velocity impact. Since then, several investigations have been carried out which have substantiated in some ways von Karman's proposed theory, and in its amended form it stands today as the authority upon which the new study is based. It does not by itself give the answers. Just as the theory of elasticity does not solve all static design problems, the von Karman work is clearly limited by

assumptions and non-ideal conditions. It will, however, serve as a welcome foundation upon which investigators can base future research projects into more specific questions of the field.

SUMMARY OF THEORY[9]

Within the elastic limit of a material, the stress in a semi-infinite bar with one end fixed whose other end is instantly subjected to any velocity V_0 can be computed from the following relationship:

$$e = V_0 t = \epsilon . l = \epsilon . c . t$$

where e = total strain

V_0 = velocity of end of bar

t = time after velocity began

ϵ = unit strain in bar

l = length of strained part of bar = distance stress wave has progress in time t.

c = velocity of stress wave = $\sqrt{\frac{E}{\rho}} = \sqrt{\frac{\text{Young's Modulus}}{\text{Mass density of material}}}$

so $V_0 = \epsilon . c$

$$\epsilon = \frac{V_0}{c}$$

and stress in part of bar being strained = $\sigma = \epsilon . E = \frac{V_0}{c} . E$

Theodore von Kármán about 1942 undertook the extension of this method of rational analysis into the plastic range by introducing the variable slope $\frac{\partial \sigma}{\partial \epsilon}$ as a substitute for E to include the plastic region of the stress strain curve. It then follows that the velocity of the plastic stress wave becomes,

$$V_\epsilon = \sqrt{\frac{\partial \sigma}{\partial \epsilon} / \rho}$$

So for the same case as above in both elastic and plastic ranges

$$e = \int_0^\epsilon V_\epsilon . t . d\epsilon \qquad \text{by analogy}$$

$$\text{so} \qquad V_0 t = t \cdot \int_0^\epsilon V_\epsilon . d\epsilon$$

$$\text{or} \qquad V_0 = \int_0^\epsilon V_\epsilon . d\epsilon$$

Then if ϵ is assigned the value corresponding to the strain at which necking begins, ϵ_m, the required impact velocity to cause abrupt failure at the moving end may be indicated.

$$V_0 = \int_0^{\epsilon_m} V_\epsilon . d\epsilon$$

Given the static stress strain curve for the material, the value of the integral can be computed. This

value V_o is then known as the "critical velocity" or that velocity at which failure will occur at the extreme end of the bar and no plastic deformation will take place beyond it.

This failure is supposedly characterized by low absorption of energy and small total elongation. It is important to note that although V_o as applied to the end of the bar is assumed constant, it may be attained in any manner and still cause failure though, in the progressive case, plastic strains may be found in regions removed from the end.

The above discussion although necessarily abbreviated, presents the principal points of the generally accepted theory governing high velocity impact. This paper presents in the discussion of results, an application of the critical velocity concept to the material being tested.

In selecting a topic for thesis work, the investigators decided that it would be a step forward to have available at the Institute, a machine with which high velocity impact investigations might be carried on by future workers. There are very few such machines in existence. It was hoped that work would be completed in time to allow

sample tests of aluminum specimens to determine what information might be obtained from them. The proper and complete study of high velocity impact must eventually include point to point determination of stress and strain in the elastic and plastic ranges of samples subjected to tensile, compressive, bending, and torsional forces. This would be a formidable undertaking. It was decided to restrict the project to tensile tests applied to conventional tensile specimens, to measure strain at several points on the specimen, and to confine instrumentation to that required for recording strain versus time. This implies neglecting energy of fracture, a characteristic considered most important in the conventional Charpy and Izod tests. However, a preliminary study of work performed on the subject indicates beyond reasonable doubt that at relatively low velocities the energy absorbed by the specimen approaches zero. The work contemplated being beyond this velocity range, it was decided to neglect energy as indicative of impact properties and to concentrate efforts upon a study of ductility, stress, and strain distribution in the specimen.

IV SPECIMENS AND MATERIAL

The material used for all of the tests was 2S aluminum. It was received as $\frac{1}{2}$" diameter bar stock and was subsequently machined to the shape shown in Fig. (1). The specimens shown with the longer ends were used for static tests and those with the short ends for the dynamic tests.

The static tests were made using a Wiehle, (30,000 lb. capacity), loading machine for measuring the load and an A-5 electric strain gage for measuring the strain. A stress vs. strain curve was plotted and the physical characteristics of the material was computed as shown in Figure (2). The material was found to have the following characteristics under static conditions:

1. Yield strength at permanent set of .0002"/in = 12,700 psi
2. Ultimate strength of 18,800 psi
3. % Elongation of 20%
4. % Reduction in area of 61.5%
5. Modulus of elasticity of 9.55 x 10^6

It was first thought that the testing procedure could best be applied to low carbon steel since it is in such common use and so readily obtainable in most any shape. It was soon learned that due to the

relatively high strength of steel it was necessary to make the testing machine much larger than it would have to be if it were used on some material with lower strength characteristics.

Plastics were then considered as a material that might be tested, but it was learned that the data that could be obtained from strain gages would be limited. This is true since the electric strain gage is only capable of indicating strain up to a certain limit, at which time it will either break or the bond between the paper and material will separate. With this in mind, it will be simple to see that when a material such as plastic with a very low modulus of elasticity is loaded it would be possible to reach the limit of the strain gage and still be well below the elastic limit of the material. For this reason plastics were not used as a material for the tests because it seemed advisable that the results of the tests should give data pertaining to the behavior of the material up to the yield and as far beyond as the strain gage would permit.

It was felt that use of 24 aluminum represented a compromise between that of steel and plastic, in that it would yield the desired type of data without requiring a large piece of testing apparatus.

V APPARATUS

A. IMPACT MACHINE

The selection of the type of apparatus used for dynamically loading the specimen was made by carefully considering several methods. The decision was first made that the machine should possess the following desirable characteristics.

1. Adequate speed control within the limits of 100 fps and 500 fps.
2. Capable of loading specimen in tension. It was believed that loading in this manner should yield results that could easily be compared to well established data on material characteristics under static loading conditions.
3. Simple in design. This was necessary since shop facilities would not permit the design of complicated parts requiring the use of high precision equipment.
4. Light in weight and requiring little space. This requirement was necessary in order that the machine may be easily transported and assembled within the confines of the laboratory.

Some of the loading methods under consideration were as follows:

a) Projectile propelled by an explosive charge.

b) Projectile propelled by compressed air as shown in Figure (2).

c) Projectile propelled by a rotating wheel as shown in figure (3).

d) Forked hammer released from a rotating member as shown in sketch (4).

In general the type of loading which depends upon a projectile for the breaking energy does not lend itself well to tension loading since the impact force must be transmitted in a direction opposite to and along the same path as the moving projectile. This has the disadvantage of requiring a complicated coupling between the point of direct impact and the end of the specimen, and furthermore, it introduces the possibility of eccentric loading on the specimen. Another chief disadvantage of the projectile is that it must be so proportioned in size that it is large enough to possess adequate breaking energy at the lower velocities and at the same time be small enough to permit its being propelled to the higher velocities using the same apparatus with limited power facilities. This suggests that a different projectile must be used for every range of velocity, all of which further complicates the apparatus. A method for the firing of a projectile at controlled speed is a difficult problem when viewing it from the standpoint of simplicity in construction. Speed control within any

degree of accuracy using an explosive charge is almost impossible to obtain and, therefore, this method was discarded as a possible choice.

The method using compressed air was considered in detail and it was found that the machine would be too large and would require too much in the way of precise machine work in connection with the fabrication of the cylinder and piston air seal.

The method using a projectile propelled from a rotating wheel was discarded because the ball projectile had to be approximately 1½ inches in diameter to obtain the desired striking energy which requires an excessively large specimen, wheel, race and driving motor.

It was finally decided that the rotating device using a forked hammer, (as shown in Figure (4)) was the method of all those considered which best suited the purpose. Detailed design calculations indicated that it could be made in one small compact unit, it could be made with a minimum of machine tools, and possessed the distinguishing features of having very good speed control and being capable of loading a specimen in tension. A simple and direct couple between the point of impact and

the end of the specimen could easily be devised by using a rectangular block hereafter called a tup, which is attached directly to the loaded end of the specimen and which receives the impact blow from the forked hammer. This is illustrated in Figure (4).

The design of the machine which was finally used was based on the premise that it would be driven by an electric, direct current motor with a wide range of speed control. A motor of this type was available which had a speed range between 100 RPM and 1800 RPM so it was decided that it could be used. This choice affected the diameter of the rotating member and, therefore, a diameter of 3 feet was used. This made the distance between the center of rotation and the center of impact equal to 18 inches. The linear relationship between the angular velocity of the member and the striking velocity of hammer was then; Revolutions per Minute x 0.1571 = fps. Using a two to one pulley connection between the motor and the rotating member allowed a possible velocity range of 39.3 fps to 360 fps.

The apparatus shown in photographs 4 to 8 was devised as the best solution to the problem from the standpoint of simplicity in operation and construction. It consisted to a 3"O.D. x ¼" wall steel tube through

which the striking hammer could slide. The mechanism for releasing the hammer consisted of a ring equipped with slots which engaged a pin mounted in the hammer as shown in photographs C and E. This ring was made with a projecting lug which engaged a tripping plunger as shown in photograph B. When the contact between the ring lug and the trip plunger was made, the rotating movement of the tube forced the ring to rotate about the tube and thus release the hammer. The hammer was to be propelled by centrifugal force along the axis of the tube until it was stopped by the pin bearing against the end of the slot in the tube. The forked end of the hammer was then in a position to clear the specimen and to engage the tup, thus imparting the impact loading that was desired. The opposite end of the tube was provided with a means of balancing the entire rotating assembly. A movable weight was used for this purpose which was actuated by screw threads and provided with a lock screw. The tripping plunger was spring loaded and was actuated by a lever as shown in photograph (C).

The problem of how the specimen could best be held in a position for striking without creating any interference with the path of the hammer was solved

by placing that end not struck by the hammer into a slot as shown in photograph F. The connection between the top and the specimen was made as shown in photograph F in order that the striking force would be transmitted from the top to the specimen uniformly, thus eliminating any possibility of specimen fracture at that point.

It seemed advisable to have a small initial load on the specimen in order that it might be properly seated in the holder slot and in the top. This was accomplished by means of a spring loaded bell crank on either side of the top as shown in photographs (E) and (F).

The rotating tube was keyed to a shaft which was mounted in ball bearings and coupled to a 1 1/4 H.P.; 110 V; 800 to 1800 R.P.M. D.C. motor equipped with a movable armature speed control. The driving couple between the armature and the shaft consisted of a "V" belt drive with a pulley ratio of 2 to 1 from the motor to the shaft. The direction of this pulley ratio was reversed at the lower speeds.

After the machine, as originally designed, had been constructed it was then subjected to a series of tests to determine just how it would stand up under operating conditions.

The machine was first tested to investigate all possibilities of misalignment and destructive vibrations. This was done by first driving the machine very slowly and then gradually increasing the speed up to 1000 RPM, taking notice of the machine's response to an increase in speed and any irregularities that occured. This test proved that the machine was properly aligned and that the rotating member was balanced well enough to reduce troublesome vibrations.

The second group of tests was made in order to determine how the tripping mechanism and the hammer would stand up under operation. This test was repeated four times at low speed ranging between 250 RPM and 400 RPM. There were no specimens fractured during these tests. These tests clearly indicated that one mechanical weakness is inherent in this type of machine; namely that the pin holding the hammer in place in the tube slot must be made quite large and be made from a high grade tool steel. The pin that was used was made from 3/8" diameter drill rod, but was subject to bending when the hammer was tripped at a speed of 250 RPM. It was also learned that the ring which holds the pin in place and which trips the hammer was not able to withstand the impact blow imparted to it when the tripping plunger was released.

It was somewhat distorted in the area around the end of the slot where the ring had driven itself against the pin. The screw shown in photograph (5) was placed in the tube in such a manner that it would bear against the tapered edge of the ring. It was believed that the ring would wedge itself between the screw and the welded shoulder on the tube and thus prevent it from turning past the point where the ring could be struck by the pin. This screw was to further serve the purpose of securing the ring in a position after impact in order that it wouldn't spin around and engage the trip plunger again.

After construction of the machine had been completed, it had virtually the same form as described above with only slight modifications. The hammer and pin were rebuilt in such a manner that the weight of the hammer was reduced by about 25% and the cross sectional area of the pin was increased by 37%. The pin was made from SAE 4150 steel, was soaked for 2 hours at 1575°F, quenched in water and tempered at 800°F for two hours. The hammer and tup were made from SAE 1045 soaked for 5 hours at 1575°F, quenched in oil and tempered at 400°F for two hours. The final shape of the hammer and pin was as shown in Figure (5). The ring was not rebuilt because it could be made to

function properly even though it was somewhat distorted as a result of the first series of trial runs. The machine was then ready for the final test with the specimen in place, the strain gages mounted, and connected to the related instruments.

V APPARATUS

B. INSTRUMENTATION Figure (7)

The problem of instrumentation presented was to devise a method of recording the strain-time relationship during impact at several points on the specimen. Electrical strain gages are the only gages available with the instantaneous response required to accurately indicate these strain changes that occur in intervals of time measured in micro-seconds. (For example, at 200 fps, a moderate loading rate, assuming instantaneous velocity attained by the end of the specimen with a 1" gage length, it would require 0.37 micro-seconds for the aluminum sample to exceed its yield strain. The actual time required is of course actually longer, but how much longer no one knows.) Of the various types of electric gages in use the resistance type was selected for its compactness, simplicity, expendability, and availability. The type CB gage was selected as the shortest (1/8") gage length commercially obtainable suited to dynamic work. A six channel General Electric strain gage amplifier received the 1000 cycle gage signal through the unbalanced voltage

of a Wheatstone bridge. A four stage amplifier increased the signal, rectified it and fed it into a six channel General Electric Oscillograph type PS-10-B2 with high speed camera attached to record the galvanometer traces. The galvanometers were single element G2 type with a frequency response of about 5000 cycles. The magazine type film holder was used to take advantage of the highest possible film speed, 1270 RPM, which corresponded to 1270 fpm or 254 inches/sec. On such a time scale each inch corresponded to 3.94 milliseconds. A Hewlett-Packard Model 202 D Audio Amplifier was used to feed a timing wave into the galvanometer at 400 cycles. The set-up as finally used is shown in Figure (7) and photograph (K).

Automatic synchronization of camera exposure and specimen rupture was a problem which was not solved and was avoided by manually opening the camera shutter just prior to applying the impact load and closing it immediately thereafter.

Careful tuning of bridge circuits was important and found to be necessary every few minutes as the bridges drifted off balance.

Each gage circuit was calibrated electrically by placing a known resistance in parallel with the strain gage and noting the displacement of the galvanometer beam.

A mechanical steam-turbine tachometer was used to measure rotational speed of the shaft upon which the rotating beam was mounted for computing striking velocities.

One important innovation in the instrumentation was the application of the strain gages directly to the sample at the forward and rear ends of the reduced section (See photograph (F)). It had been the custom in the past to mount the gages on an adjacent section which transmitted the impact load to the specimen. This practice saved strain gages and did indicate the load being applied to the end of the specimen, but it gave no indication of the strain and stress distribution during impact in the specimen itself in terms of position relative to point of load application. It was hoped that direct measurement of strain-time curves of points on the specimen would provide a basis for analysis of non-ideal impact. A third C-5 gage was mounted on the holder of the fixed end of the specimen in an effort to determine correlation of readings on this and the direct mounted gages.

VI TEST PROCEDURE

1. The strain gages were cemented to the sample at the desired points and, after drying, were mounted in the tup and fixed holder.
2. The gage circuits were completed using soldered connections to shielded leads, taking care that leads were clear of hammer.
3. The bridge circuits were balanced with amplifier phase and amplitude adjustments starting at lowest attenuator setting and working up to #10 gain setting so that minimum equilibrium current was flowing.
5. All gage circuits were calibrated with known resistance in terms of micro-inches/inch per mm. of galvanometer beam deflection. Used 5 ohm galvanometer input resistance.
6. With camera motor up to speed the timing wave was recorded by use of transient shutter control.
7. Impact machine was started and brought up to desired speed of 360 RPM as indicated by tachometer by adjusting armature speed control on motor. On this run 90 VDC was used to energize the motor from a motor-generator converter set.
8. Lamp voltage on oscillograph was set to maximum

value, camera shutter opened manually, trigger release on machine was tripped to initiate impact blow, and camera shutter was closed. This was done in succession as rapidly as possible.

9. Lamp voltage reduced. Motors stopped.

10. Gage length and reduced diameter measured. Temperature of room noted and film record developed and printed.

VII IRREGULARITIES IN TESTING

The behavior of the loading machine during all of the tests was observed and the following mechanical weaknesses were found to exist:

1. The pin which is mounted in the hammer and which slides along the slot in the tube was subject to a brittle fracture close to the center. This failure took place at the highest testing velocity of 189 fps. The fracture took place at a point where a small quench crack existed. It is believed that this weakness was largely a result of fully hardening the pin.

2. The tripping ring was subject to distortion during every test. This weakness is plainly a result of too soft a grade of steel and poor design. The stop that was provided was not adequate nor was the ring made to sufficiently large dimensions. The very nature of the ring's function makes a rational design exceedingly difficult.

3. The location of the tripping plunger was

such that the hammer was released in such a position after tripping that it struck the top of the tup before making a complete revolution. Evidence that this occurred could be seen when the specimen was inspected, since large marks were found in the top end where the tup had gouged into the specimen as a result of an irregular loading condition. This is also evidenced in the photographic oscillograph trace (photograph J), by the small undulating signal which diminishes in magnitude as if the specimen were struck and allowed to damp out its own vibration. It will be noted that this trace is the same on both channel #1 and #2 which received the signal from the gages on the specimen. This irregularity was a result of poor design since the plunger could well have been placed beyond the top to avoid this difficulty.

4. The assembly which was subject to rotation at high velocity caused considerable noise due to wind resistance and it was noticed that the motor's speed control wasn't very sensitive at the higher speeds due to this heavy load imparted to it. It is believed that the lug on the tripping ring was largely responsible for the wind friction encountered.

The ring lag could well have been made a little shorter.

5. The top-end of the specimen was subject to some bending at the velocity of 187fps mostly due to the blow it received when it was caught by the box of cotton waste and rags. The box about 6 inches deep, wasn't deep enough to cushion the blow properly.

6. The ends of the slots in both the tube and the hammer were subject to a small amount of upset as a result of the impact received when the hammer was thrown out. This could have been minimised by providing a cushion at the end of the slot.

7. The bearing support channel to which the top was attached was subjected to more lateral thrust than was anticipated due to the ring's movement about the tube. The channel was forced outward enough to cause some clearance between the shoulders on the driving shaft and the bearing race. This is serious because the tube assembly might move far enough in a lateral direction to cause the specimen holder to be struck by the hammer.

8. Elimination of random pick-up in the circuit was not satisfactory as evidenced by the 1000 cycle signal impressed on the recorded signals (photograph 8). Fully shielded and grounded leads were

used and all instrument chassis were grounded, however, it is felt that the prepared plug-in leads used were excessively long.

9. The manual tripping of the camera shutter is not a satisfactory method of control since it caused over-exposure of the zero and open circuit lines on the record. It was necessary, due to inability to devise a satisfactory synchronizing method for camera exposure in the time allowed.

10. Under-voltage on the lamp in the oscillograph may have been the cause of the faint rupture track obtained on the film. This was due to a shorted rheostat control on the oscillograph for which no spare was available.

11. The timing wave failed to appear on the film record due to the width of the film being less than anticipated. It was believed that the film would record the full length of the visual screen, about 8 inches, whereas it actually extended only over 5 3/4 inches at the left end of the screen.

VIII DATA

Sample A - STATIC TEST

As Gage Factor = 2.01 Sample Dia. = 0.252"

Load	Gage		Total	(psi)
0	S-722			0
50	825	103	103	1000
100	1033	108	211	2000
150	1145	112	323	3000
200	1252	107	430	4000
250	1362	110	540	5000
275	1412	50	590	5500
300	1463	51	641	6000
325	1518	55	696	6500
350	1570	52	748	7000
375	1624	54	802	7500
400	1680	56	858	8000
425	1736	56	914	8500
450	1797	61	975	9000
475	1860	63	1038	9500
500	1925	65	1103	10000
525	6-1001	76	1179	10500
550	1073	72	1251	11000
575	1151	78	1329	11500
600	1236	87	1416	12000
625	1350	92	1508	12500
650	1458	108	1616	13000
675	1640	102	1718	13500
700	1676	136	1854	14000
725	1839	163	2017	14500
750	7-1038	199	2216	15000
775	1288	250	2466	15500
800	1650	362	2828	16000
825	8-1218	568	3396	16500
850	10- 190	962	4358	17000
875	-1950	1770	6128	17500
900	46-1400	9430	15578	18000
916	A 10-2000+	11000+	26578+	18300
925	Failure	-	-	

Gage Length Initial = 1.000"
Final = 1.203"

Diam. Initial = 0.252"
Final = 0.109"

ELECTRICAL CALIBRATION:

R (Calibrating Resistance) = 161,000

(12)

$$\epsilon_c = \frac{R_g}{GF\,(R_c + R_g)}$$

For C-8 gages

$$GF = 2.86$$

$$R_g = 500$$

$$\epsilon_c = \frac{500}{2.86\ (161,500)} = 1082 \text{ micro-in./in.}$$

For C-8 gage

$$GF = 2.86$$

$$R_g = 362$$

$$\epsilon_c = \frac{362}{2.86\ (161,362)} = 671 \text{ micro-in./in.}$$

Channel	Amplifier Attenuator	Galvanometer Resistance (Ohms)	Scale (mm) a	b	Difference (mm)(b-a)	μ"/mm.
A	#10	5	-70.0	-12.0	+57.0	19.0
B	#10	5	-61.8	- 1.8	+60.0	21.7
C	#10	5	-20.0	+21.5	+41.5	16.2

DYNAMIC TESTS

Specimen	Gage Lt. (in.)	Reduced Diam. (in.)	Hammer Velocity (fps)
C	1.33	0.078	87
B	1.50	0.079	132
A	1.50	0.071	109

RESULTS

(a) The machine constructed delivered the desired impact tensile load at velocities up to 189 fps.

(b) Instrumentation used yielded a photographic trace of strain in the specimen before and during impact at 57 fps as shown in photograph (3).

(c) Elongation and reduction of area at various velocities for 2S aluminum are noted below:

Specimen (see photo. 3)	Velocity of loading (fps)	Nature of fracture
A	0 (static)	ragged concave
C	57	perfect concave
D	132	perfect concave
E	189	perfect concave

% Elongation	% R.A.
20	81.5
33	90.0
30	90.0
40	92.0

Sample Calculations:

Specimen 'B' (a) % elong. = $\frac{1.20-1.00}{1.00} \times 100 = 20\%$

(b) Initial Area = $\frac{(.252)}{4}$ = 0.05

Final Area = $\frac{(.109)}{4}$ = .0094

% RA = $\frac{0.05 - .0094}{0.05} \times 100$ = 81.3%

IX DISCUSSION

Photograph (4) is a print of the film record obtained of impact at 87 fps. Traces A, B and C are the zero positions of the forward and rear gages mounted on the specimen and the gage mounted on the holder, respectively. Traces D and E indicate the final position of A and B after fracture while the shutter was still open. A faint trace can be seen of the path of beam A from its original position to position D. The wave superimposed upon traces is 5000 cycle pick-up from the bridge-exciting oscillator. The timing wave was off the film to the right and would have appeared at the right margin of the film if it had been realized that the film did not extend to the entire width of the viewing screen. Fortunately the 5000 cycle corresponds to .0002 sec. of elapsed time. The distance from trace A to D represents 896 micro-inches/inch of strain and that from B to E represents 1226 micro-inches/inch. The significance of these figures is problematical. They may represent open circuit values of

amplifier current, or, if the bond between gage and specimen failed before the circuit was broken, they may represent residual strain in the gage after the impact load was applied. At any rate it can be seen that the gage was recording less than 900 micro-inches of strain a full .001 of a second after impact although at 57 fps the hammer is traveling 0.68 inches in that time, or more than the total elongation observed. It would seem that either the gage is giving a false reading or rupture is taking place at considerably below 57 fps. The latter is the more likely explanation. A third alternative is the possibility that galvanometer response is not rapid enough to accurately trace the signal. The inertia of the mirror, however small, is finite and with an extremely rapid signal it may tend to spread out the signal along the time scale.

It is possible that too high a value of amplification was used thus recording only the first part of the actual trace before coming to the limit of amplifier output. Decreasing gain would sacrifice accuracy of readings, however, and this is a factor to be considered.

Neither trace B nor C give a visible trace due possibly to a faulty rheostat on the oscillograph which limited voltage that could be applied to the lamp source of light. It is not certain that trace C, the gage on the steel holder, would have given any noticeable reading under any condition.

The calculations given in Figure (9) and shown graphically in Figure (10) result from an application of the von Karman theory to the calculation of critical velocity for the 2S aluminum. This value is about 35 fps. According to the theory summarized earlier, at impact velocities greater than 35 fps, fracture should be characterized by small total elongation. Figures (11) and (12) show that experimental data obtained at higher hammer velocities shows greater elongation than under static conditions. This indicates again the possibility that a considerable difference may exist between hammer velocity upon impact and true velocity of the end of the sample. It is obvious that the velocity of the end of the sample must accelerate from zero to an undetermined value at fracture so that

the purely theoretical case of instantaneous attainment of velocity can only be approached, and analyses of results must be made with this in mind. The exact relationship of hammer velocity and sample velocity is the unknown here. Much more work is needed to clear this up.

Also of interest is the location of the point of fracture on the sample tested at 139 fps. (Specimen 3, Photo I). The fact that it broke away from the moving end and has strong evidence of double-necking might possibly be explained by the reflection and super-position of stress waves. The double-necking phenomena has been reported in previous work.[1]

X Conclusions

The authors are aware that definite conclusions cannot be based upon the small number of tests that have been performed. However, the results of this investigation seem to indicate that the following facts may be used as a basis for further investigations.

1. The basic principle of the loading machine used in this investigation is sound and may be satisfactorily applied to impact testing at velocities up to at least 200 feet per second.

2. The use of an electrical resistance type strain gage, together with a General Electric Type PH-10-32 Oscillograph for recording strain during impact is probably limited to velocities under 100 feet per second.

3. Aluminum in the 24 form is more ductile when loaded at hammer velocities up to 189 fps than it is under static conditions.

XI SUGGESTIONS FOR FUTURE WORK

Suggestions for future work in the field of high velocity impact are divided into two groups, namely, one group which includes suggestions pertaining to modifications of the testing procedure and apparatus used in this investigation, and another group which relate in a general way to further experimental projects in the subject.

A. Suggestions for improvement of the existing apparatus are as follows:

1. Make the ring, hammer, pin, tripping plunger and tup of the highest grade tool steel available and make certain that the finished product is heat treated to yield the proper balance between high strength and toughness.

2. Redesign the hammer, pin and ring to provide for the terrific loads imparted to them. Detailed design of these parts as shown in Figure (5) is recommended for trial.

3. Place tripping plunger assembly as far forward as possible to avoid tripping hammer too soon.

4. Provide lateral bracing for the bearing support

on top side to which the tripping plunger assembly is attached. This bracing will carry the thrust caused by the tripping ring as it turns about the tube.

6. Shorten instrument leads to minimum length and ground at both ends, to eliminate pick up. Possibly isolated oscillators may improve situation.

8. Devise an electrical trip for the hammer actuated by the oscillograph transient control switch to synchronize automatic camera exposure with impact, or use continuous record film.

7. Handle strain gage leads with extreme care to eliminate grounding on sample holder or interference with hammer. Scotch tape will hold leads in place.

8. A small D.C. generator mounted on one end of the main shaft of the impact machine could be used to give a continuous record of rotating speed on the film if properly calibrated. This might be useful in investigating hammer-tup speed relationship, or in determining energy values.

9. Use 8 inch width film for recording to allow maximum amplitude for signals.

10. Consider extending length of sample by moving stationary holder away from tup. Would make it possible to use larger gage loading area, would facilitate mounting specimen by making it unnecessary to pass gages through slot in holder, and would make it easier to detect wave phenomena by increasing distance between gages.

11. Mark a grid upon reduced section of sample and measure variations in spacing after impact to obtain better overall picture of plastic flow in different parts of the section.

12. Use timing curve of a frequency to give 20 cycles per inch of record film.

13. Consider stepping up pulley ratio from camera motor to camera to obtain speed greater than 1270 ft. per minute with cascarine film.

14. Use super-sensitive film to obtain more positive track.

15. Use following settings on instruments:
(a) Attenuator setting 5 or 6 on amplifier to keep strain record within limits of film. On steel mounted gage use maximum gain.

(b) Equilibrium current less than 40 ma to avoid signal saturation by amplifier.

(c) Resistance setting 6 Ω on galvanometer to decrease pick-up.

(d) Voltage about 10 V or less during exposure to insure visible track.

8. FUTURE WORK

1. Consider a rotating device with a fixed hammer that clears the fixed specimen until a movable tab springs into position to cause the hammer to engage the specimen. This was used as described in reference (10).
2. Try cathode ray oscilloscope to obtain most sensitive response and use high speed motion picture camera for recording record.
3. Record gage signals on magnetic tape or wire recorder and play back at reduced speed for film record.
4. Experiment with brittle lacquer, determine its qualitative possibilities in the region of brittle fractures.
5. Use high speed motion picture camera on specimen during impact to determine mechanism of fracture.
6. Devise mechanism to record velocity of tup versus time during fracture. Might be done by using coil on tup and a strong magnetic field around it, or by electro - magnetic gage to indicate displacement versus time. This would give a positive record of true velocity of end of specimen.

XII SKETCHES, DIAGRAMS

and PHOTOGRAPHS

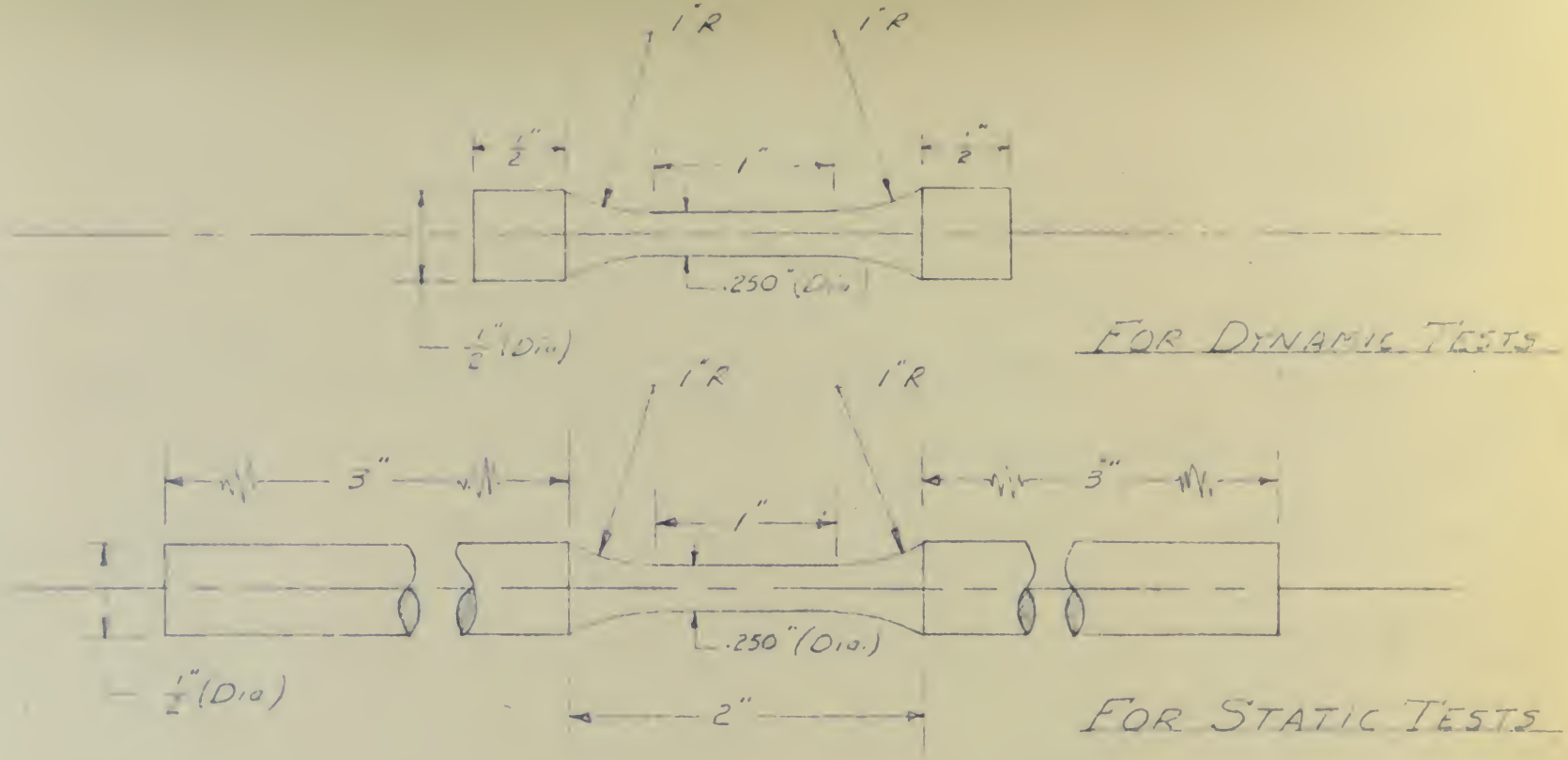


FIG 1

SPECIMENS

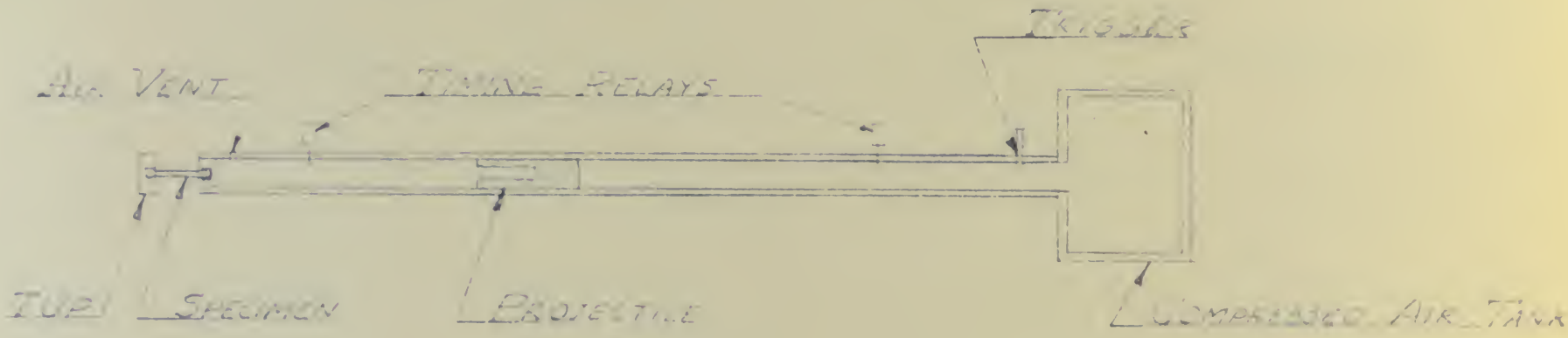


Fig 2

A Scheme for Impact Loading

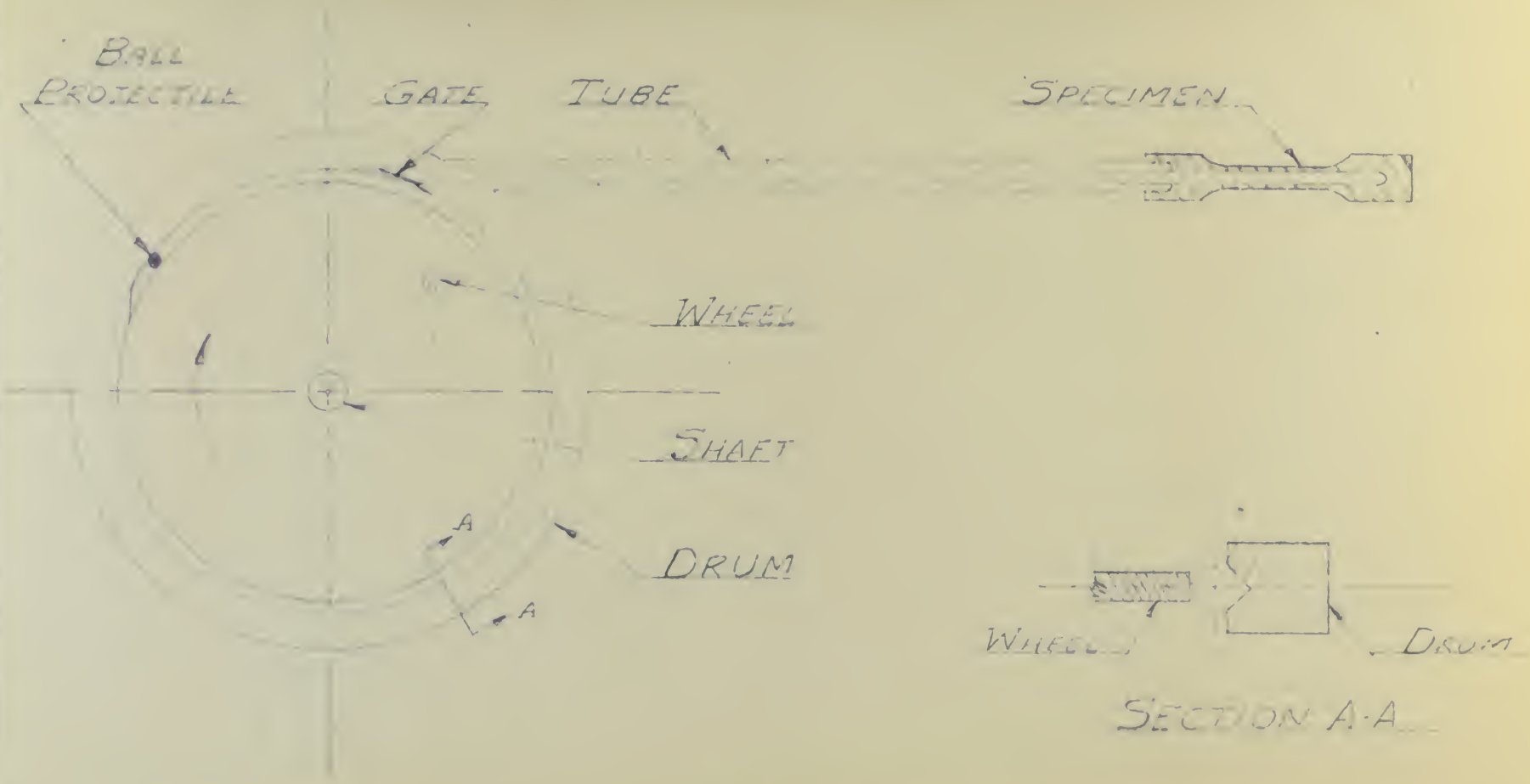


FIG. 3

A SCHEME FOR IMPACT LOADING

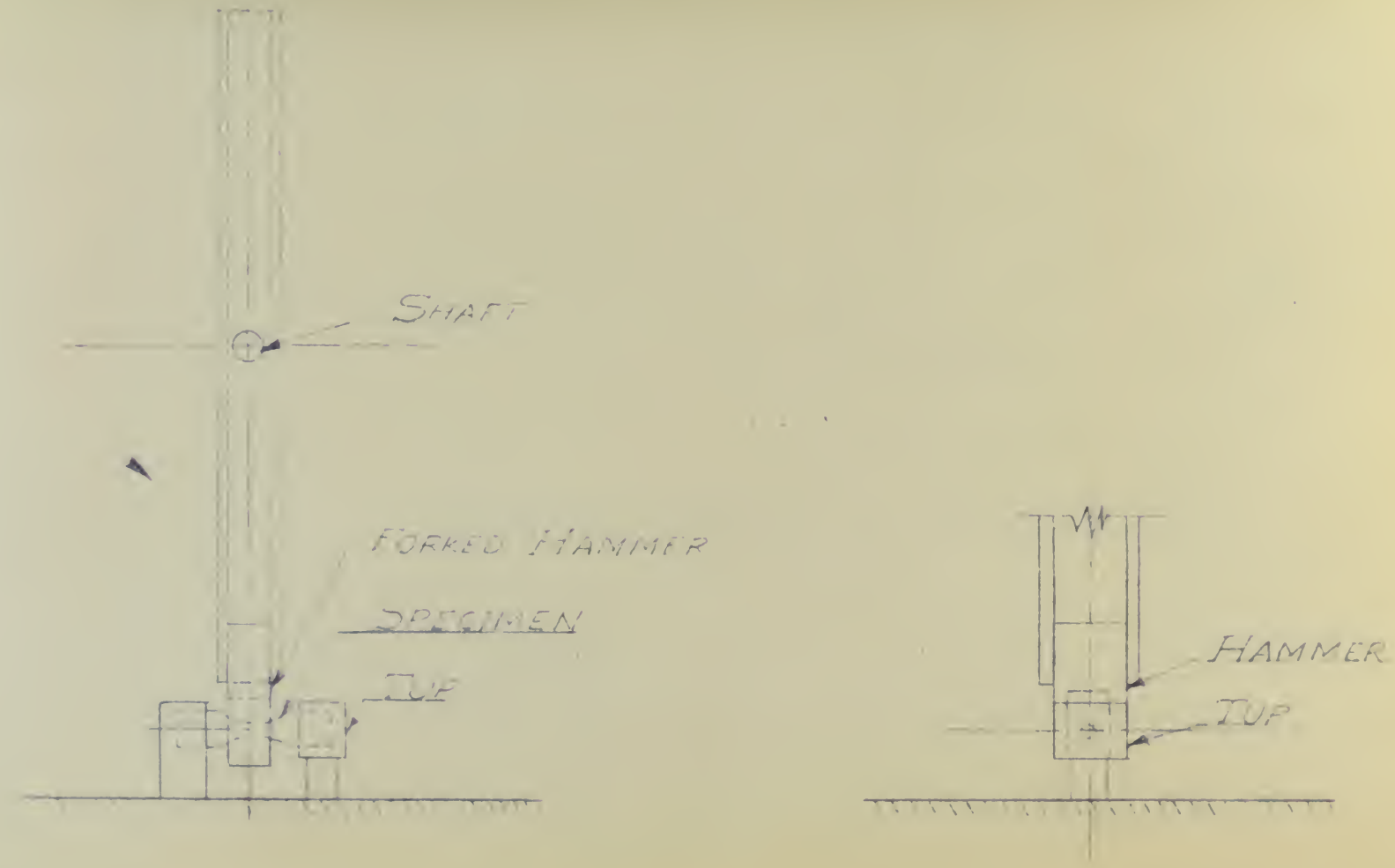
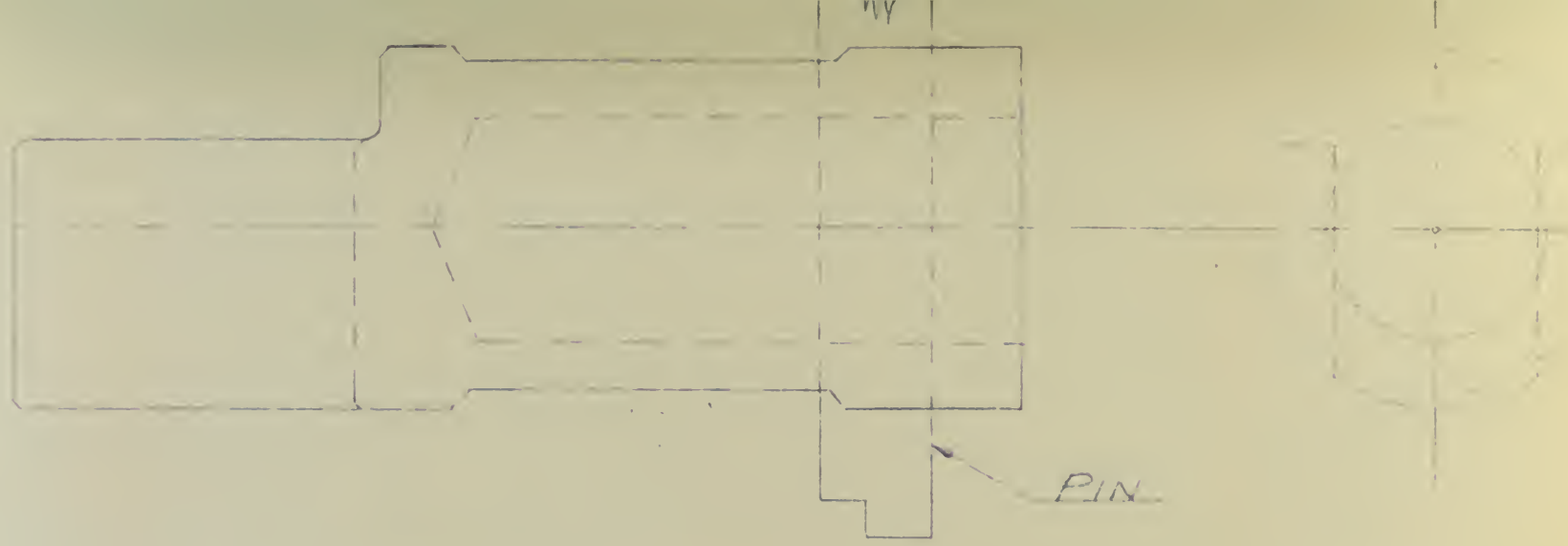
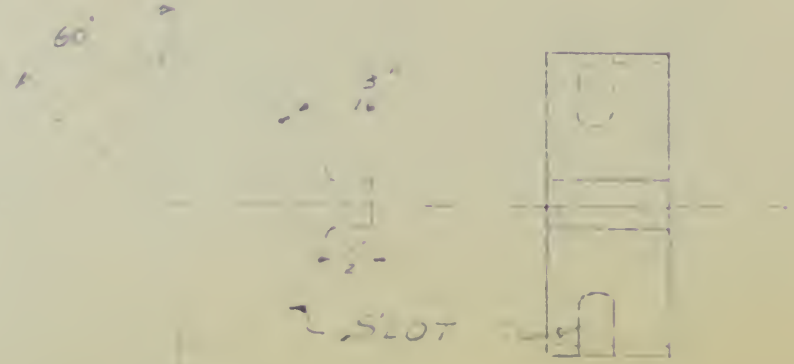


FIG. 4

A SCHEME FOR IMPACT LOADING



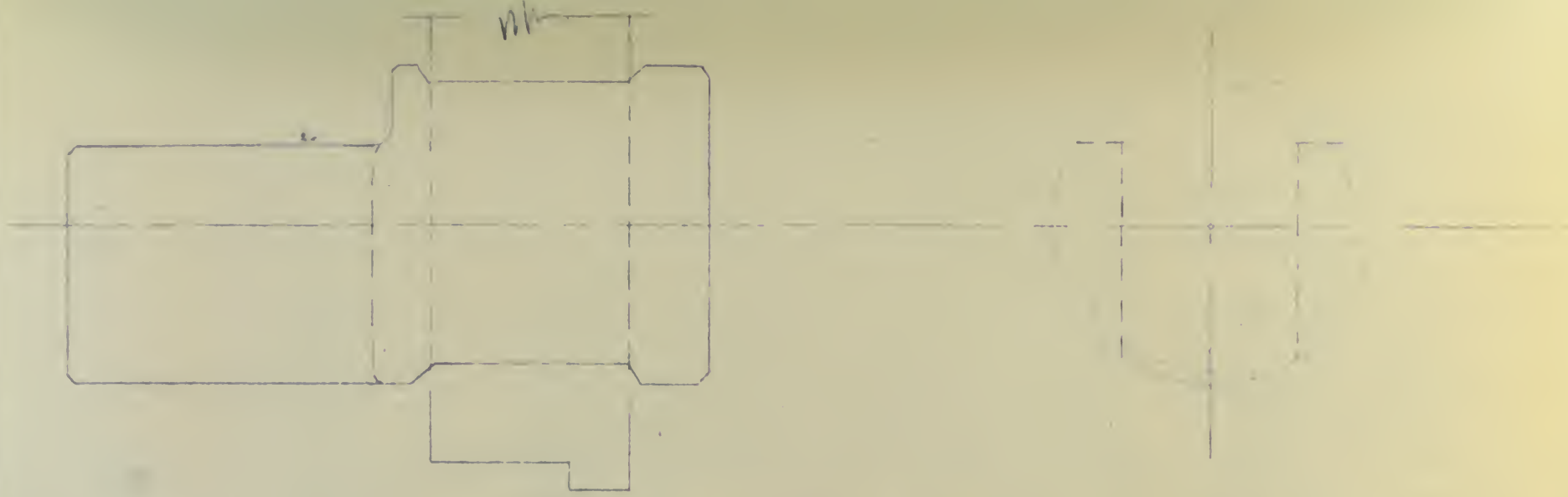
HAMMER & PIN - FULL SCALE



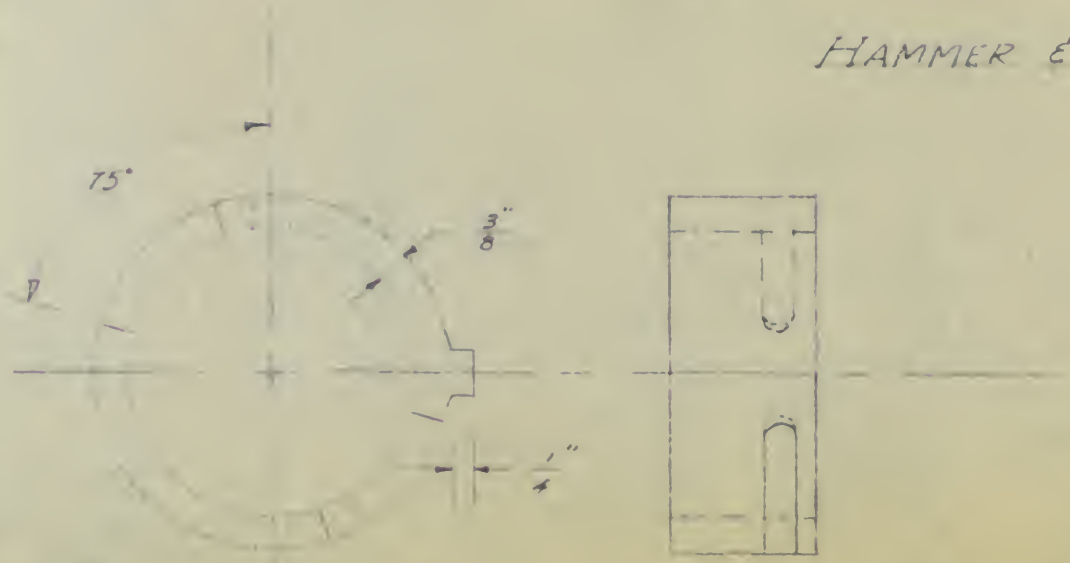
RING - HALF SCALE

FIG. 5

DETAILS OF LOADING MACHINE



HAMMER & PIN - FULL SCALE



RING - HALF SCALE

FIG. 6

PROPOSED MODIFICATION
OF LOADING MACHINE

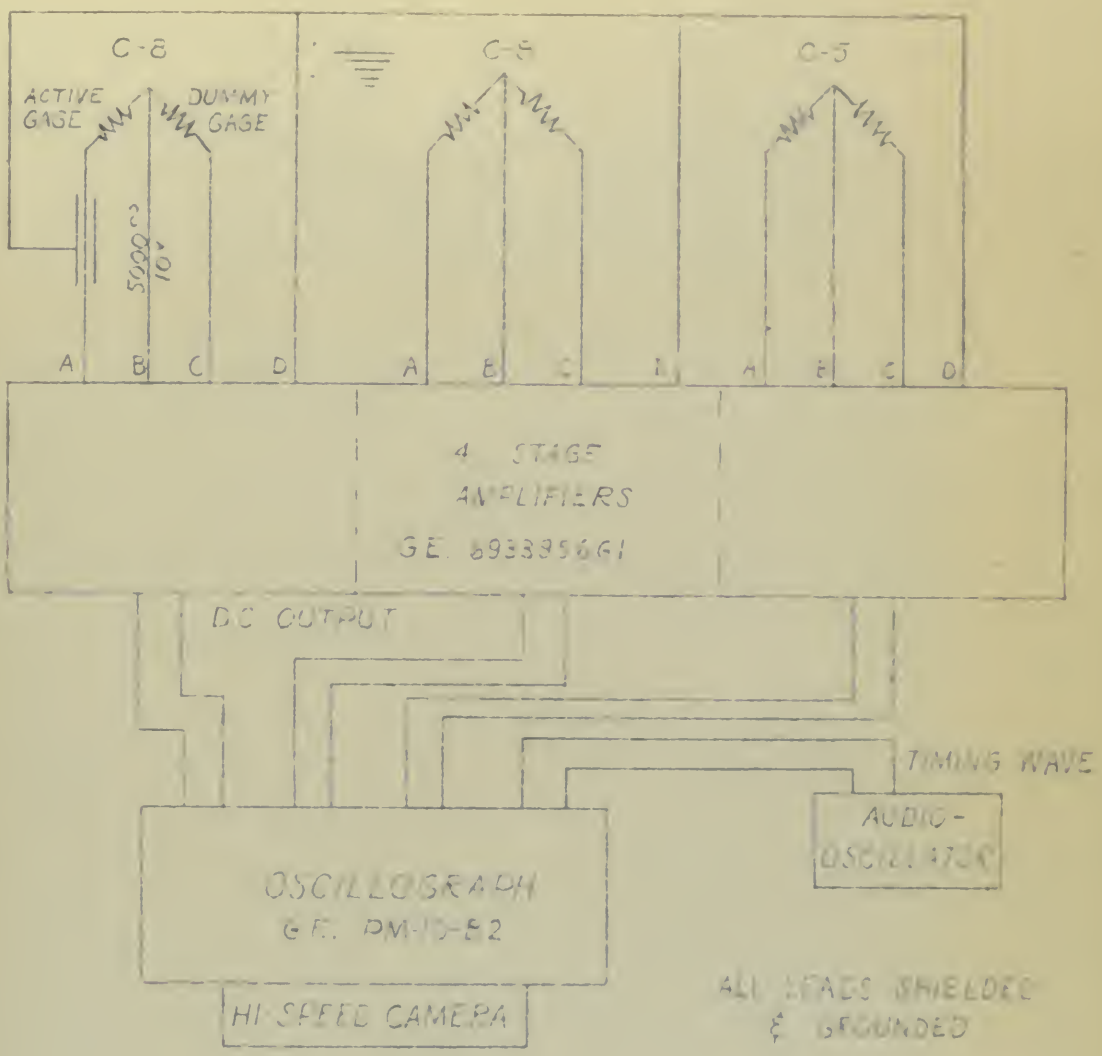


FIGURE "2" - INSTRUMENTATION

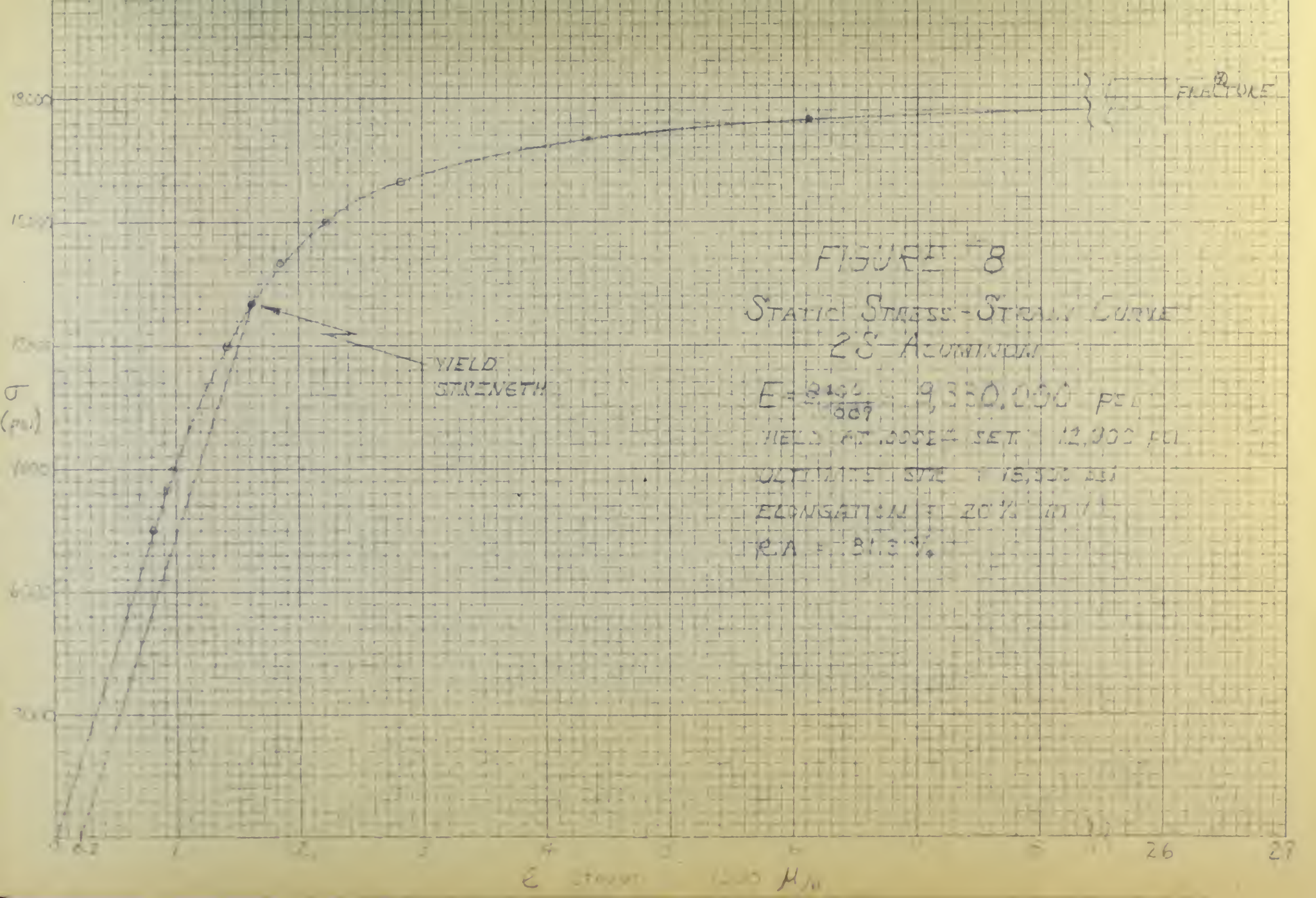


FIGURE 8

STATIC STRESS-STRAIN CURVE

2S-ALUMINUM

E = 9,350,000 PSI

YIELD AT .002 SET 12,000 PSI

ULTIMATE STR 15,000 PSI

ELONGATION IN 2" 20%

R.A. 81.3%

FIGURE #9

V_ϵ vs ϵ Curve

Calculations

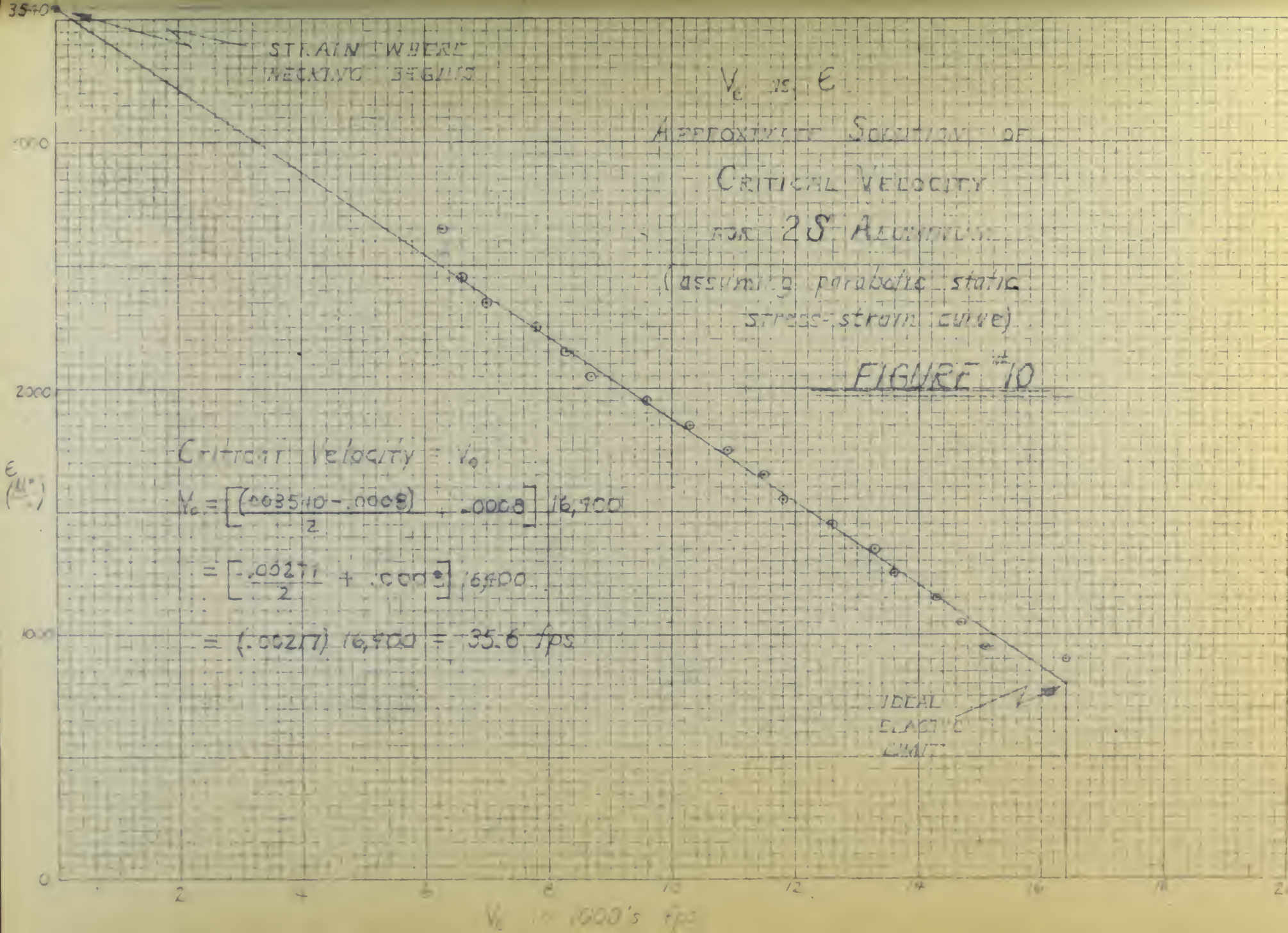
$V_\epsilon = \sqrt{\frac{d\sigma}{d\epsilon}/\rho}$. Where $\rho = 1.60$ lb/ft^3 = 5.0 lb/32.2 ft/s^2 = 4.97 lb ft^{-4}

$V_\epsilon = \frac{1}{223}\sqrt{\frac{d\sigma}{d\epsilon}}$

From ϵ	To ϵ =	$\Delta\epsilon$	$\frac{d\sigma}{d\epsilon}$ lb/ft^2	$\sqrt{\frac{d\sigma}{d\epsilon}}$	V_ϵ	$V_\epsilon \cdot \Delta\epsilon$
0	900×10^{-6}	900×10^{-6}	1343×10^{6}	36.7×10^{3}	16.4×10^{3}	14.81
900×10^{-6}	1000	100	1139	33.8	15.1	1.51
1000	1100	100	1080	32.9	14.7	1.47
1100	1200	100	1008	31.8	14.3	1.43
1200	1300	100	922	30.4	13.6	1.36
1300	1400	100	879	29.7	13.3	1.33
1400	1500	100	792	28.2	12.6	1.26
1500	1600	100	691	26.3	11.8	1.18
1600	1700	100	663	25.8	11.5	1.15
1700	1800	100	590	24.3	10.9	1.09
1800	1900	100	533	23.1	10.3	1.03
1900	2000	100	401	21.5	9.6	0.96
2000	2100	100	374	19.3	8.7	0.87
2100	2200	100	346	18.6	8.3	0.83
2200	2300	100	302	17.4	7.8	0.78
2300	2400	100	245	15.6	7.0	0.70
2400	2500	100	215	14.7	6.6	0.66
2500	2600	100	196	14.0	6.3	1.39
2600	-	-	negligible	-	-	-

Σ = 34.31

Critical Velocity = 34.3 fps


3540
STRAIN WHERE NECKING BEGINS
V_c vs ϵ
HYPOTHETICAL SOLUTION OF CRITICAL VELOCITY FOR 2S ALUMINUM
(assuming parabolic static stress-strain curve)
FIGURE 10
Critical Velocity = V_c
$V_c = \left[\frac{(.0354 - .0008)}{2} + .0008\right] 16{,}700$
$= \left[\frac{.0346}{2} + .0008\right] 16{,}700$
$= (.0181)\ 16{,}700 = 302$ fps
IDEAL ELASTIC LIMIT
2000
1000
0
ϵ (in/in)
2
4
6
8
10
12
14
16
18
20
V_c in 1000's fps

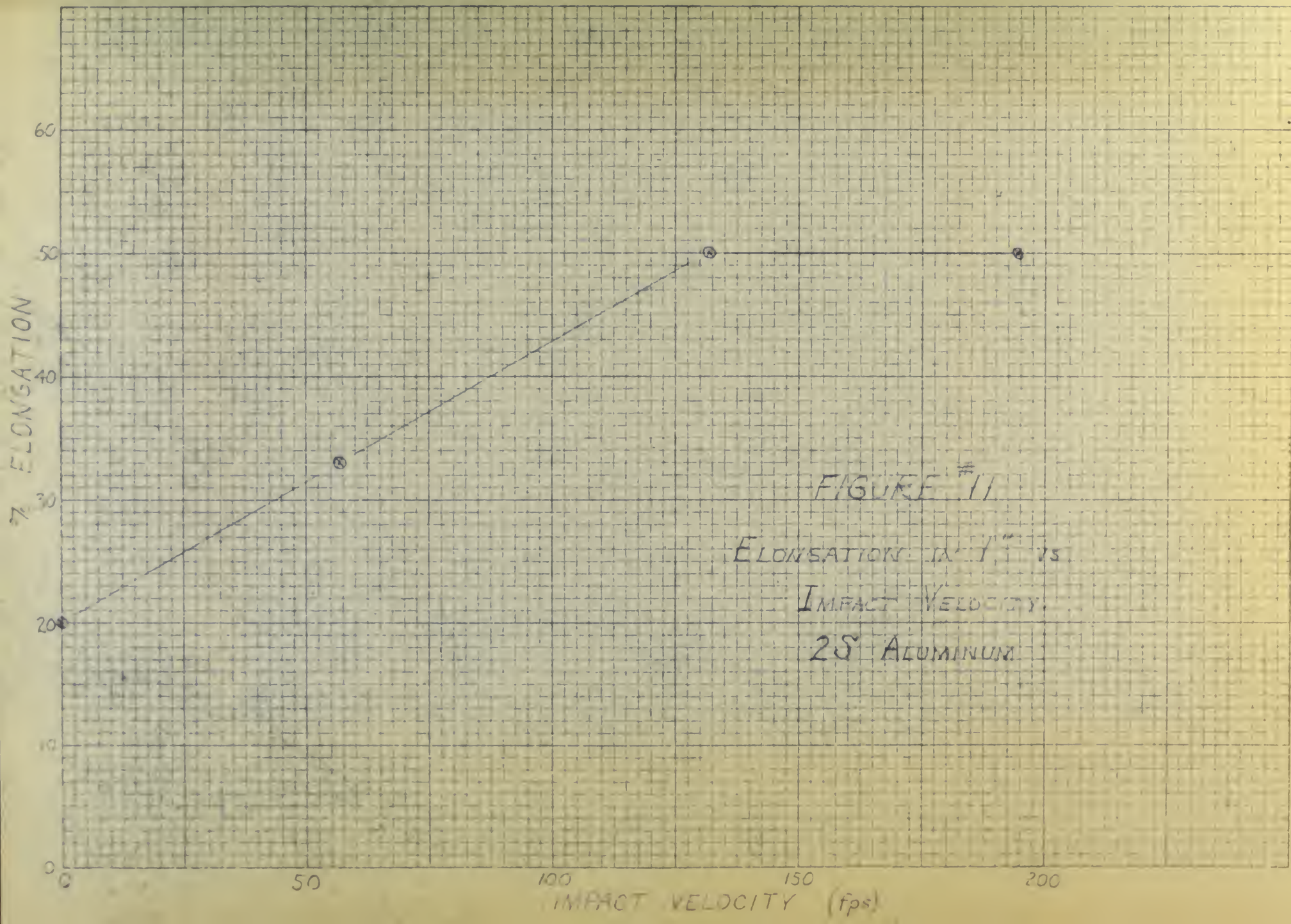


FIGURE #11

ELONGATION IN 1" VS.

IMPACT VELOCITY.

2S ALUMINUM

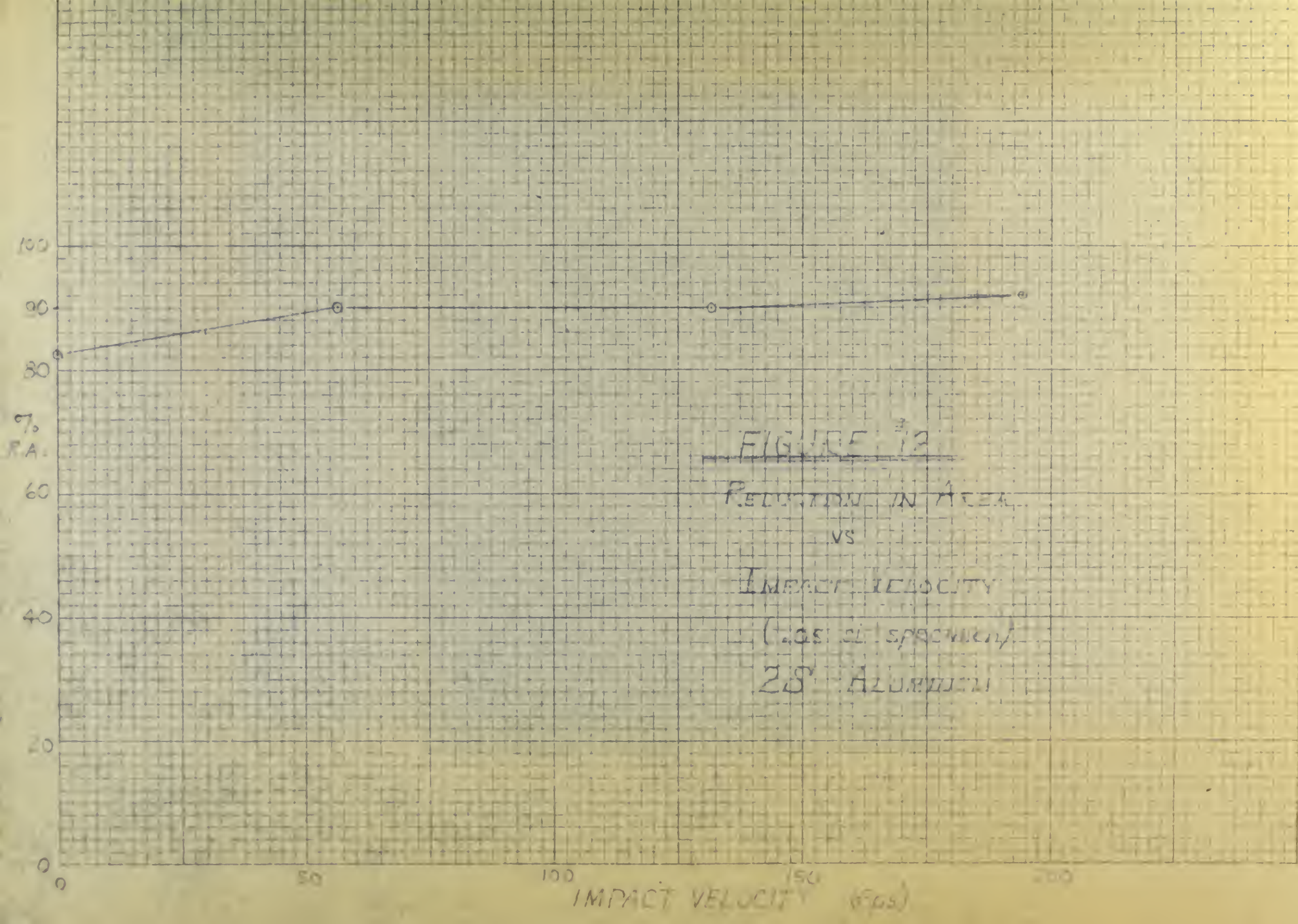


FIGURE 12

REDUCTION IN AREA

VS

IMPACT VELOCITY

(.25 CL SPECIMEN)

2S ALUMINUM

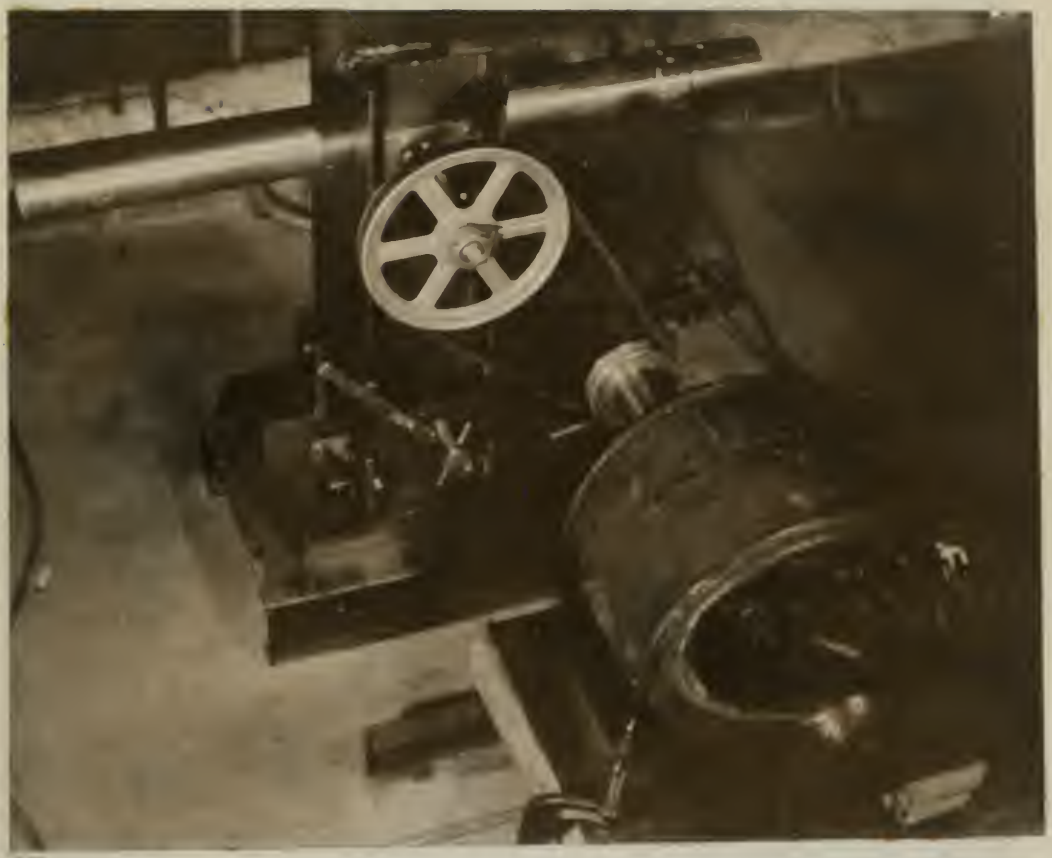


PHOTO A
General view of Loading Machine

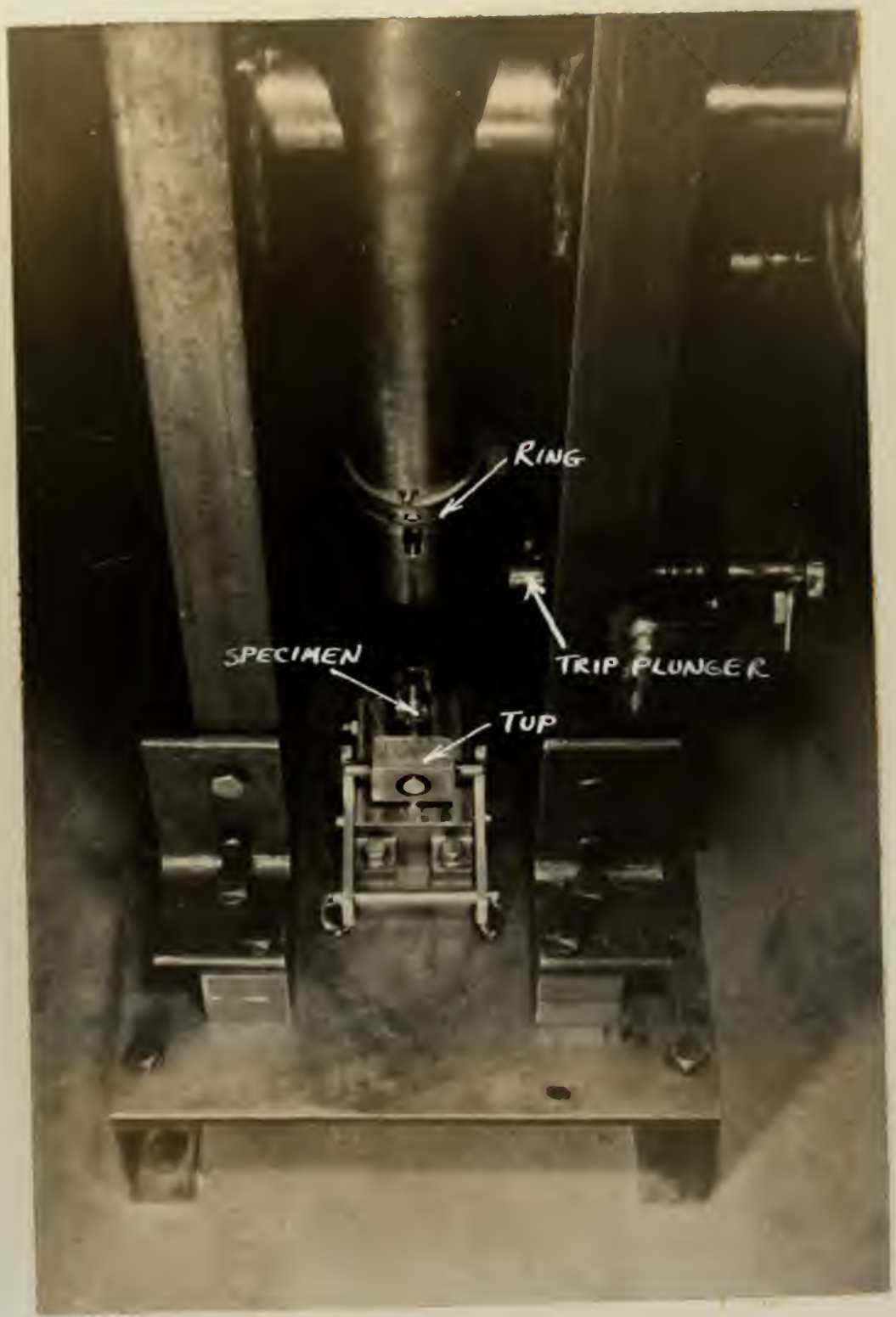


PHOTO B
Front view of Loading Machine
showing tup, tube and hammer

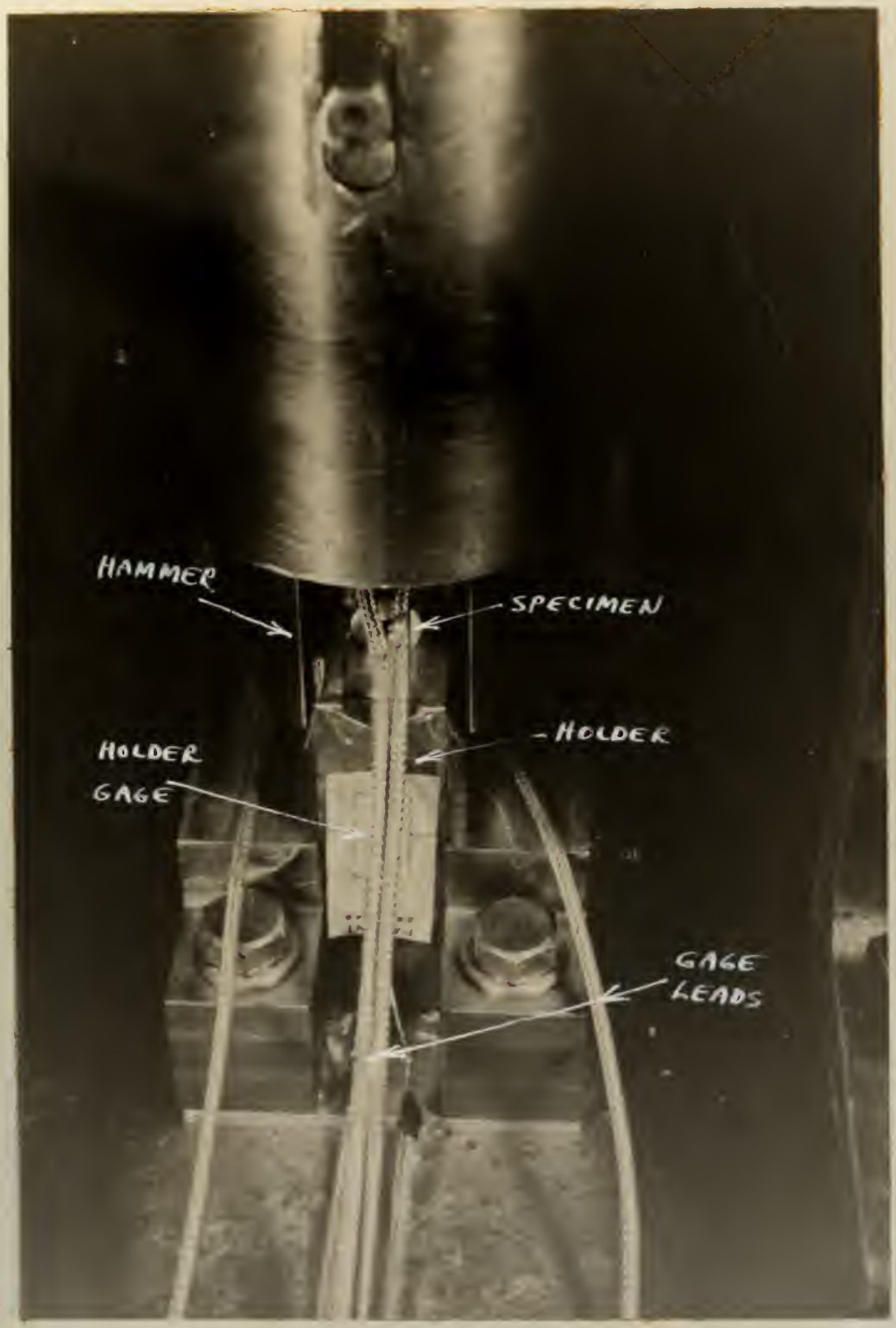


PHOTO C
Rear view of Loading Machine
showing tube, hammer, holder and gages

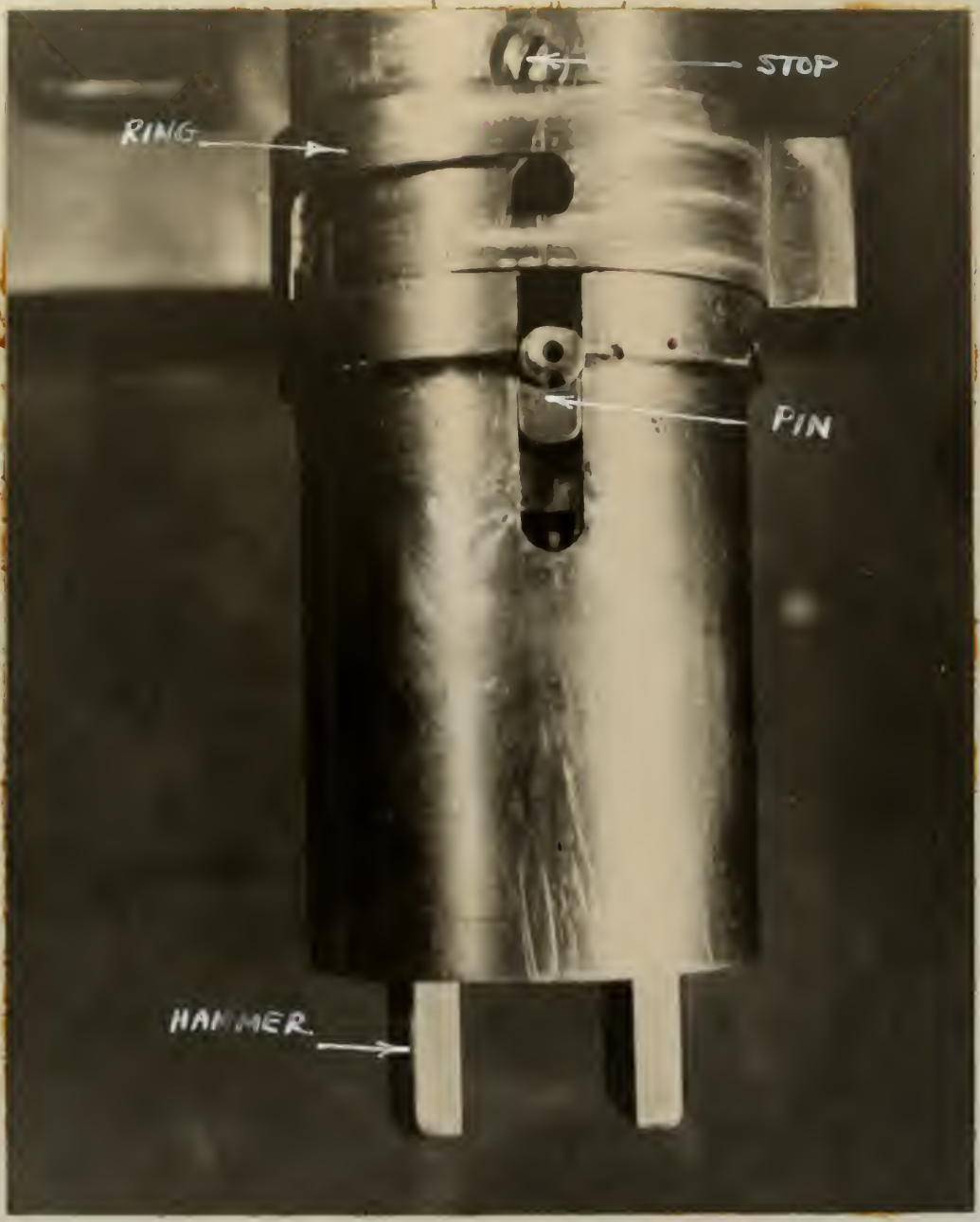


PHOTO D
View of Hammer and Ring

PHOTO E
View of Hammer and Ring

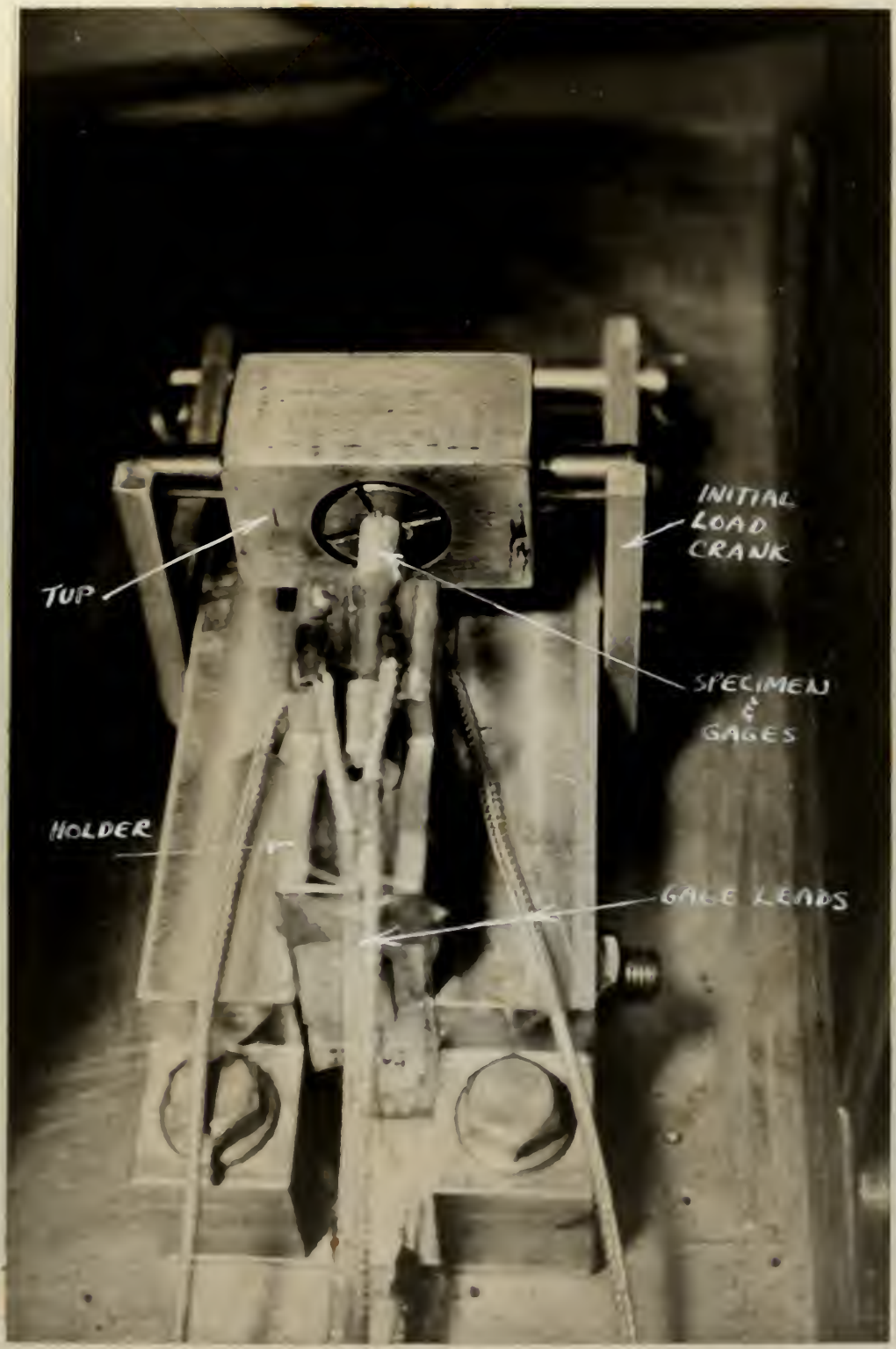


PHOTO 9
View of Tup, Holder and Specimen

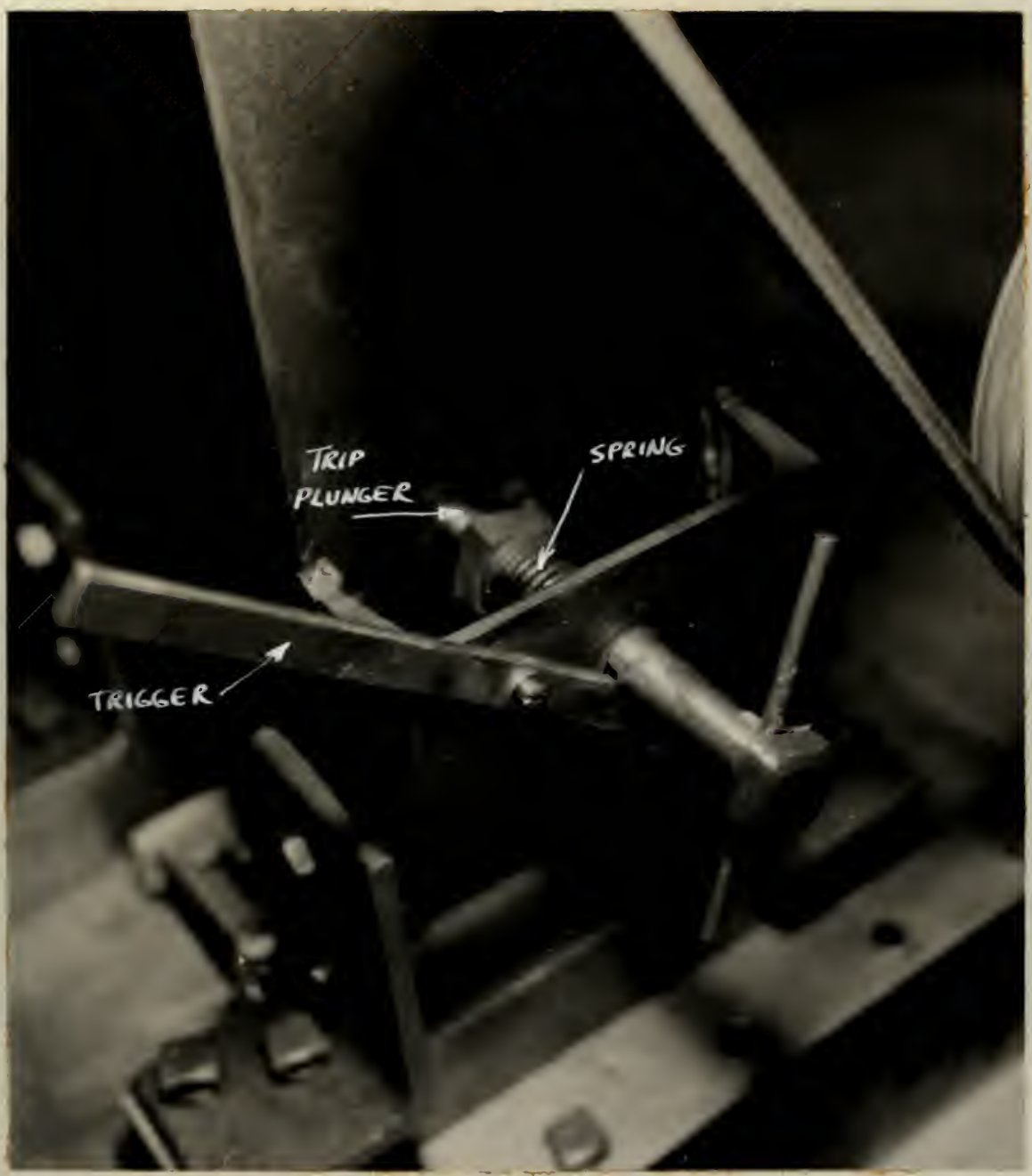


PHOTO G
View of Tripping Plunger

PHOTO B
View of Instruments

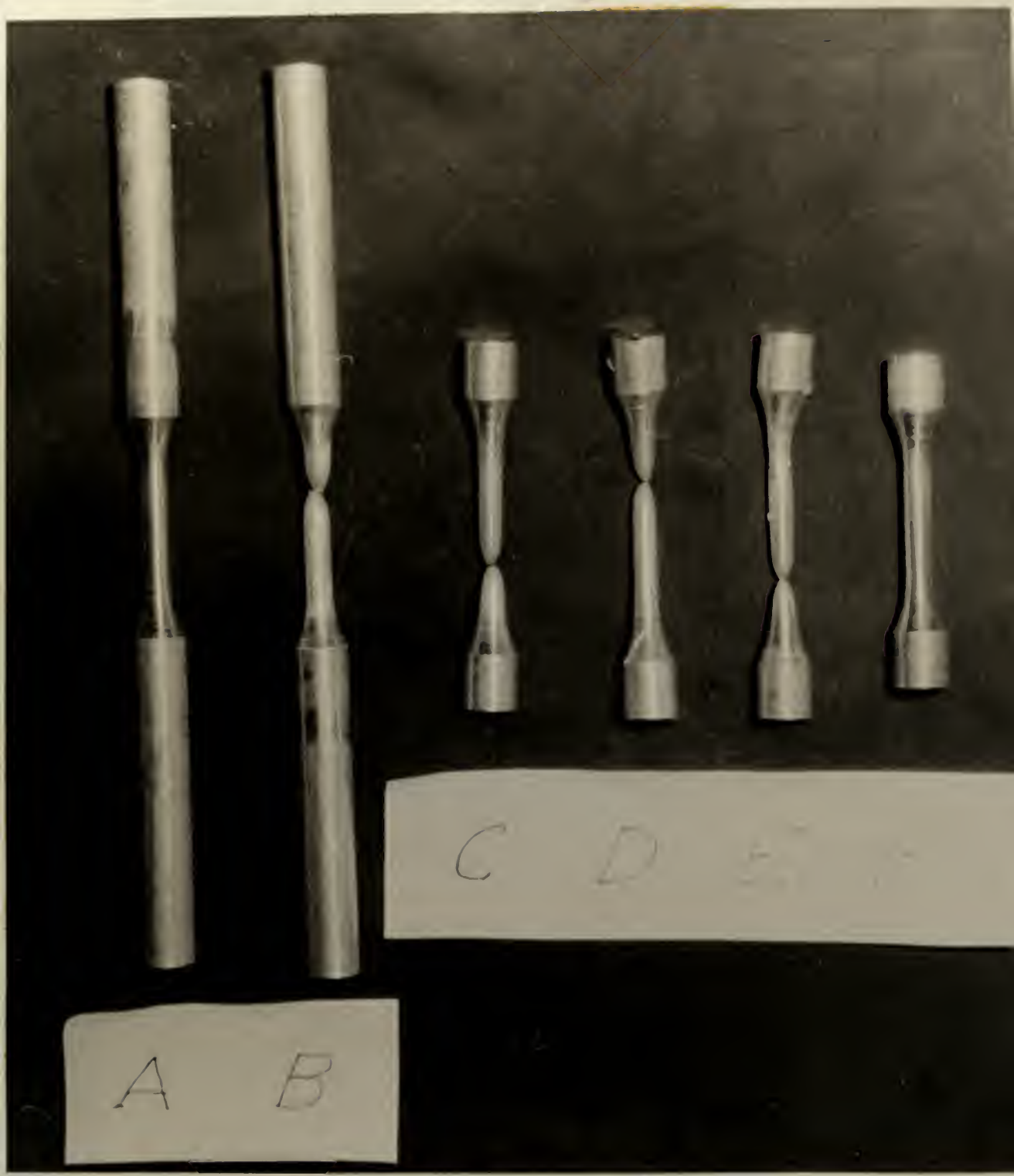


PHOTO I

Specimen A - Static Before Testing
" B - " After "
" C - Impact - Fractured at 67 fps
" D - " - " " 132 "
" E - " - " " 189 "
" F - " - Before testing

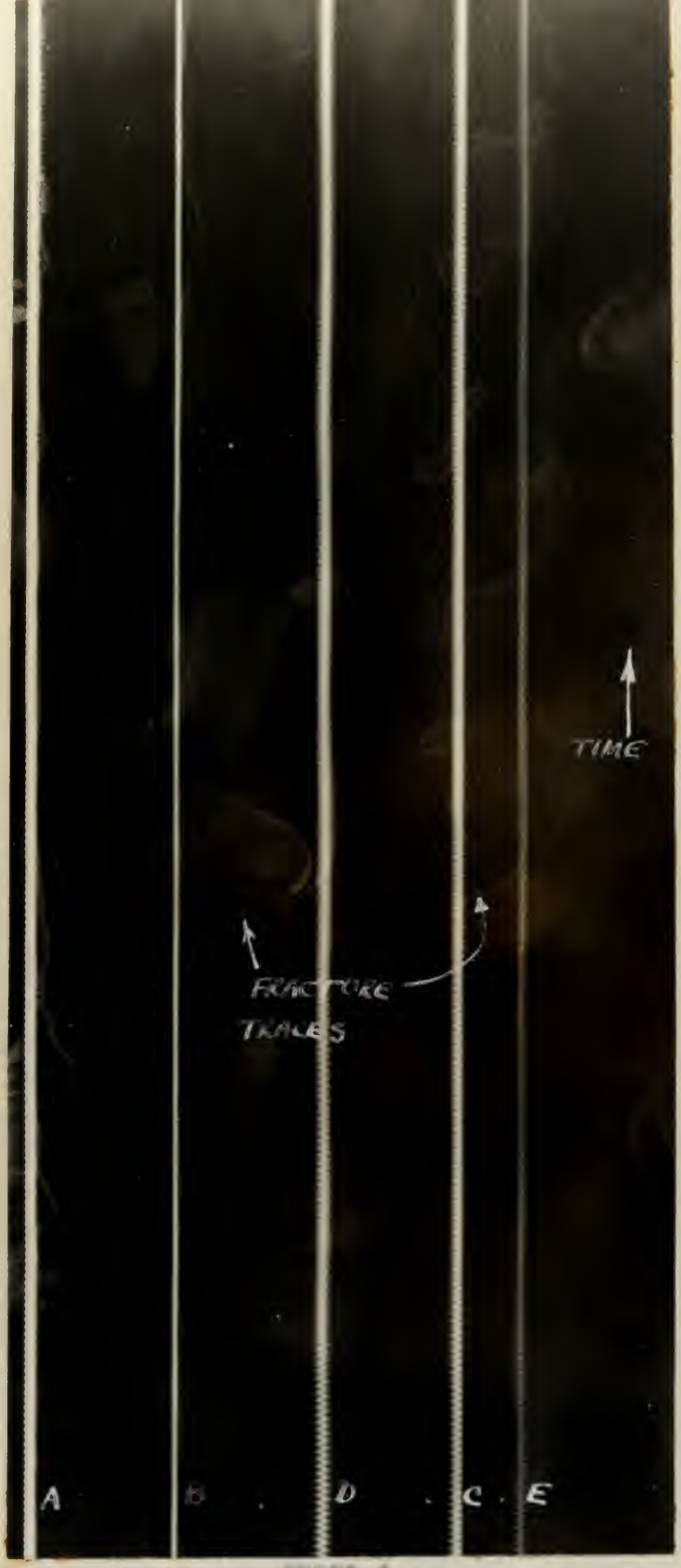


PHOTO J
Oscillographic Record

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Thesis 7475
S48 Sholders
High velocity impact.